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EFFECTS OF LEAKAGE THROUGH CLEARANCE SEALS ON THE PERFORMANCE OF A 10 K STIRLING-CYCLE REFRIGERATOR

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The use of clearance seals is essential to achieve long-life, wear-free operation of Stirling-cycle cryogenic refrigerators. This paper describes an experiment which determined the effect of leakage through clearance seals on the performance of such a refrigerator operating at temperatures ranging from 20 K down to 10 K.

The ability of a Stirling-cycle refrigerator to achieve 10 K with clearance seals was successfully demonstrated. Results indicate that the leakage flow undergoes gap regeneration before reaching the cold expansion volume. A simple model of gap regeneration was used to estimate the regeneration loss due to the leakage flow. This regeneration process minimizes the loss in refrigerator performance caused by the clearance-seal leakage. As a result, clearance seals remain effective down to a refrigeration temperature of 10 K.

Key words: clearance seal; cryogenic refrigerator; cryogenics; heat transfer; regenerator; Stirling cycle.

1. Introduction

One of the life-limiting mechanisms in a typical cryogenic refrigerator is the wear of its seal surfaces. Wear of the seal surface not only increases leakage but also generates contaminants. In order to achieve long-life, wear-free operation of cryogenic refrigerators, the use of clearance seal is essential.

A clearance seal is a long, narrow annular gap established between the outside surface of a reciprocating cylinder and the internal surface of a mating cylindrical housing. Sealing is attained by the flow resistance provided by the long narrow gap. Clearance seals were first implemented successfully on a long-life, Stirling-cycle refrigerator operating down to 30 K [1]. In this study, a triple-expansion, Stirling-cycle refrigerator was tested to study the effectiveness of clearance seals to attain temperatures below 20 K. This paper describes the experiment with the three-stage refrigerator and a first order analytical model of the effects of the clearance seal on cold production.

2. Description of refrigerator

The first successful operation of the triple-expansion, Stirling-cycle cryogenic refrigerator used in this study was reported by Daniels and duPré [2] in 1971. In this earlier study, this refrigerator originally reached a temperature of 9 K with a third-stage (coldest) regenerator filled with lead spheres. The refrigerator configuration is typical of conventional Stirling-cycle machines driven by a simple crank-type mechanism (fig. 1). The compression heat is rejected to water through a cooling jacket. The crankshaft is driven externally by a variable-speed motor.

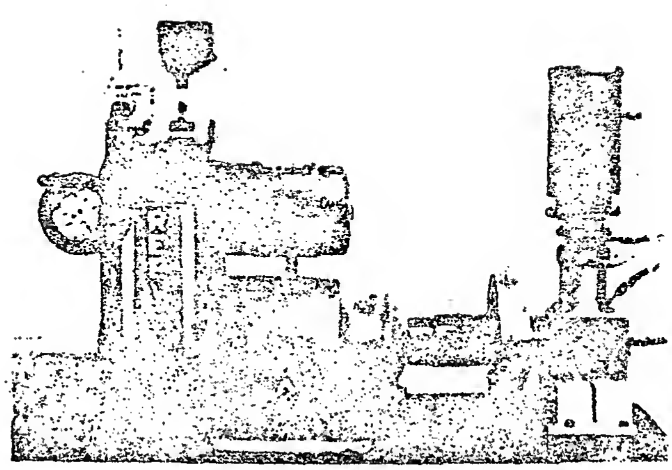

Figure 1. Motor drive and triple expansion refrigerator

The displacer is stepped to three diameters (figs. 2 and 3), with each section containing a regenerator matrix. The first (i.e., warmest) and second stage matrices are made of layers of phosphor-bronze mesh.

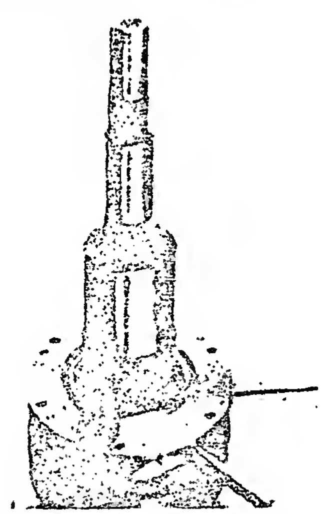

Figure 2. Three stage displacer

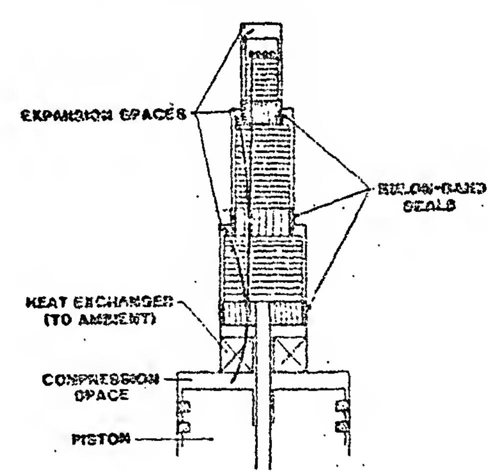

Figure 3. Schematic of triple expansion refrigerator

The original lead spheres were replaced by lead mesh. Lead mesh is made by expanding lead-calcium-tin alloy sheets (fig. 4). Lead mesh was used as a test for its potential as an alternative to lead spheres.

Each of the three displacer stages has a Rulon-band seal (see fig. 3) which is epoxied to the lower section, and was machined to fit their respective bores in the cold finger. The seals also act as bearings guiding the reciprocating motion of the displacer.

The temperature of the coldest stage was monitored with a helium gas thermometer. The 1st and 2nd stage temperatures were monitored with copper-constantan thermocouples mounted on the flanges of the cold finger. An electric resistance heater was mounted to the third stage to measure the cold production.

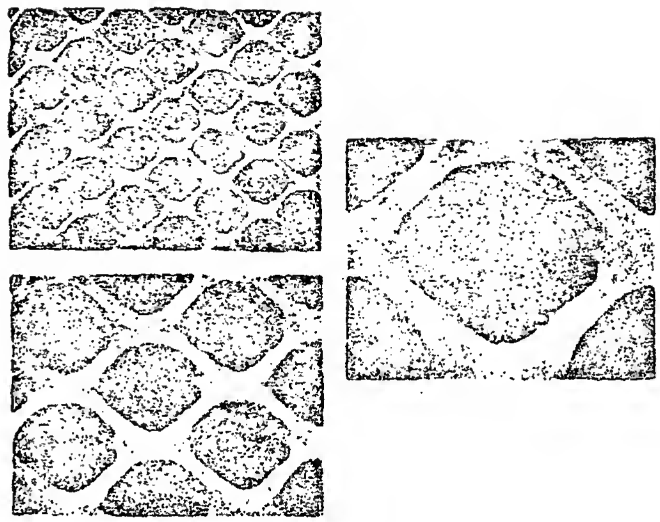

Figure 4. Magnified view of lead mesh

The refrigerator was first run with all contacting seals and a temperature of 9.4 K was reached. Table 1 summarizes the major dimensions and operating conditions of the refrigerator.

Table 1. Major refrigerator dimensions and operating conditions

	1st stage	2nd stage	3rd stage
Displacer diameters, mm	39.9	20.0	15.1
Regenerator:			
length, mm	39.4	29.4	28.5
mesh material	phosphor-bronze	phosphor-bronze	lead
mesh fill-factor	0.4	0.4	0.4
Piston diameter, mm	63.5		
Displacer stroke, mm	12		
Piston stroke, mm	32		
Displacer-piston phase angle	60		
Charge pressure, psia	89.6		
Speed, cps	640		
Piston swept volume c.c.	100.8		

3. Tests with clearance seal

After a temperature below 10 K was achieved with all contacting seals, the diameter of the third stage seal was reduced to provide a radial clearance seal. The seals/bearings on the second and third stage were responsible for guiding the displacer. The total indicated runout of the clearance seal was less than 0.0002". The refrigerator was re-run with a third-stage clearance seal of 0.0015" radial clearance. The test was then repeated with a 0.002" radial clearance seal.

4. Simplified model of gap regeneration

Working fluid leaking through the clearance seal flows past a long, narrow annular gap before reaching the expansion space (fig. 5). The leakage flow undergoes gap regeneration by exchanging heat with the two concentric walls surrounding the annulus. The loss in cold production due to clearance-seal leakage is a result of imperfect gap regeneration experienced by the leakage flow.

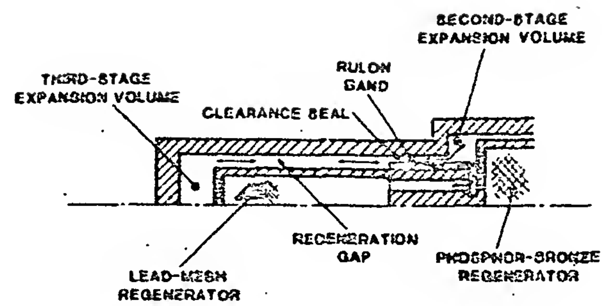

Figure 5. Schematic of leakage flow passing through the clearance seal and regeneration gap with dimension of the clearances exaggerated

A model was developed to estimate the regeneration loss of the leakage flow.

In order to treat the involved regeneration theory analytically, the following simplifications are made.

1. The temperature and pressure of the fluid in the regeneration gap vary spacially with axial position, z, only.
2. The time-averaged wall temperature varies linearly with z.
3. The thermal properties of both fluid and wall are constant.
4. The regeneration losses are the sum of the losses from two independent cases. In case 1, the regenerator has finite heat transfer rate but infinite wall heat capacity. In case 2, the regenerator has finite wall heat capacity but infinite heat transfer rate, and the fluid pressure fluctuates with time.

Case 1. Finite heat transfer

The leakage with an average mass flow rate of $\dot{m}_0$, flows through an annular regeneration gap with average diameter D and gap width s. If heat transfers between the fluid and wall through a coefficient h, the energy equation is:

$$\dot{m}_o c_p \frac{d\,T_g}{d\,z} = 2\pi D h (T_g - T_w), \qquad (1)$$

where T_g = fluid temperature

T_w = wall temperature

c_p = specific heat of fluid

The heat transfer coefficient for flow through an annulus with equal linear temperatures on both walls is given by [3]

$h = 8.23 \frac{k}{2s}$, where k = thermal conductivity of fluid.

With a linear wall temperature, eq. (1) can be integrated over the entire seal length L to yield

$$T_g(L) = T_c + \frac{T_h - T_c}{\Lambda} \; (1-e^{\Lambda}) \; . \tag{2}$$

where T_h = temperature at the warm end of the regeneration gap,

T_c = temperature at the cold end of the regeneration gap,

and $\Lambda = \dfrac{2\pi dLh}{\dot{m}_o c_p}$

For laminar flow in the gap, the average mass flow rate in the gap can be expressed as

$$\dot{m}_o = \frac{\pi\rho D d^3 \Delta p}{12\sqrt{2}\,\mu\ell} \; , \tag{3}$$

where ρ = fluid density,
μ = fluid viscosity,
d = clearance seal width,
ℓ = clearance seal length,
and
Δp = pressure drop along the entire seal length.

The regeneration loss due to finite heat transfer, q_h is then given by

$$q_h = \dot{m}_o c_p (T_g(L) - T_c) \; ,$$

or using eq. (2),

$$q_h = \dot{m}_o c_p \frac{T_h - T_c}{\Lambda} (1-e^{-\Lambda}) \; . \tag{4}$$

Case 2. Finite heat capacity and fluid pressure fluctuation

The wall heat capacity available for regeneration is limited by the thermal penetration depth, δ, of the wall material. Since heat transfer in or out of the wall over half of the cycle, δ is given as

$$\delta = \left(\frac{\alpha_w}{\omega}\right)^{1/2} \; ,$$

where α_w = thermal diffusivity of the wall
ω = angular frequency of the refrigeration cycle

Thus the wall heat capacity per unit length available for regeneration is $2\rho_w c_w \pi D\delta$, where ρ_w and c_w are density and specific heat of the wall material respectively.

Since the effects of finite heat capacity and fluid pressure fluctuation are time-varying, the transient energy equation must be used to analyze the effects. Applying the energy equation on a volume of fluid contained in a length of dz of the regeneration gap, one has [4]

$$-2\rho_w c_w \frac{\delta}{s} \frac{\partial T_w}{\partial t} = \rho c_p \frac{DT_g}{Dt} - \frac{DP}{Dt}, \tag{5}$$

where P is fluid pressure. If flow resistance is neglected and the assumption of infinite heat transfer (i.e., $T_w = T_g$) is used, eq. (5) can be written as

$$\frac{\partial T_g}{\partial t} = \frac{1}{c_m} \frac{\partial P}{\partial t} - \frac{\dot{m}}{\pi D s} \frac{c_p}{c} \frac{\partial T_g}{\partial z}, \tag{6}$$

where $c_m = \rho c_p + 2\rho_w c_w \frac{\delta}{s}$.

The time-varying pressure and mass flow rate can be expressed as

$$P = P_o - \hat{P} \cos \omega t \tag{7}$$

$$\dot{m} = \hat{m} \cos (\omega t - \theta) \tag{8}$$

$$\hat{m} = \sqrt{2}\, \dot{m}_o .$$

The fluctuation of pressure and mass flow in the regeneration gap are modeled as being in phase with the fluctuation in the main working volume.

Using eqs. (7) and (8), one can integrate eq. (6) to obtain

$$T_g = \frac{P}{c_m} - \frac{\hat{m}}{\pi D s \omega} \frac{c_p}{c_m} \frac{\partial T_g}{\partial z} \sin (\omega t - \theta) . \tag{9}$$

The regenerator loss due to finite heat capacity and pressure fluctuation, q_c, is given by the rate of flow of enthalpy, which is

$$q_c = \frac{\omega}{2\pi} \oint \dot{m} c_p T_g dt$$

$$= \frac{\omega}{2\pi} \frac{c_p}{c_m} \oint \dot{m} P dt ,$$

or

$$q_c = \frac{\hat{P}}{2} \frac{c_p}{c_m} \hat{m} \cos \theta . \tag{10}$$

The total loss due to clearance seal leakage is given by the sum of q_h and q_c.

5. Results and Discussion

Cold production of the triple-expansion refrigerator at temperatures ranging from 10 K to 20 K were measured. In figure 6, the top curve shows the performance of the refrigerator with line-to-line contacting seal, and the lower two curves represent the 0.0015" and 0.002" clearance seals. The differences between the top curve and the lower two are the losses in cold production due to the leakage of the clearance seals. These measured losses are plotted in figure 7 together with the losses predicted by eqs. (4) and (10).

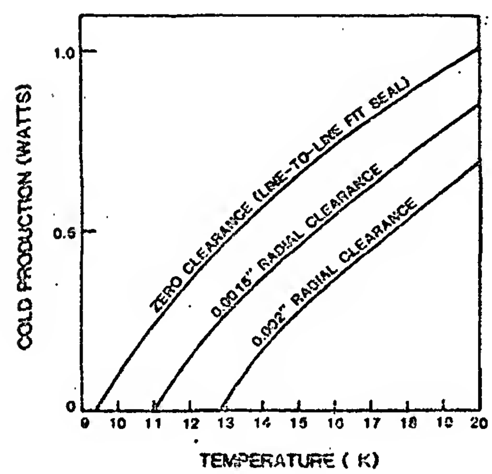

Figure 6. Cold production at the coldest stage

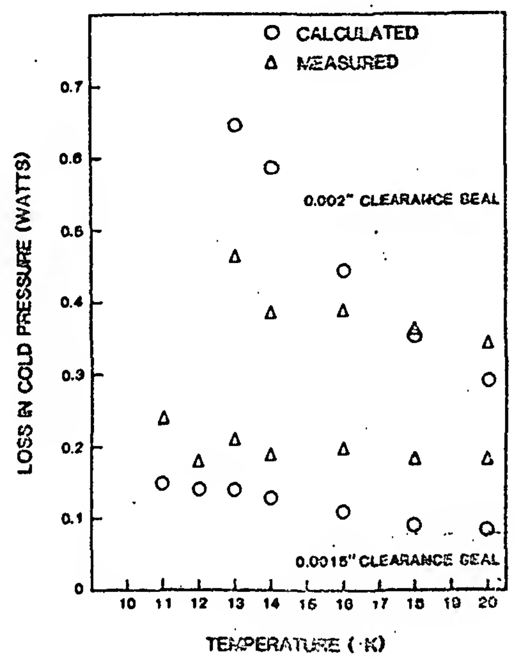

Figure 7. Loss in cold production due to clearance seal leakage

The discrepancy between the measured and calculated results is most likely due to the simplification made in the theoretical model for gap regeneration. Furthermore, friction heating by the contacting seal and slight eccentricity of the annular clearance seals were not accounted for in the model. But the comparisons show that eqs. (4) and (10) give good first order estimation of the loss due to the use of clearance seals in Stirling-cycle refrigerators.

This experiment shows that the clearance seal losses produce a warm-up of 1.6 K and 3.5 K with clearances of 0.0015" and 0.002", respectively. This is an acceptable penalty in the design of a refrigerator with long-life as the leading priority. Furthermore, the loss due to clearance seals can be significantly reduced by using seals with smaller clearances. At the present, refrigerator with clearance of 0.001" is in operation [1], and the design of refrigerator with clearances of 0.00075" is in progress [5].

6. Conclusions

Measurements from a triple-expansion Stirling refrigerator show that the loss caused by clearance seals is low and is an acceptable penalty in the design of a long-life cryogenic refrigerator. The losses due to clearance-seal leakage can be modeled as imperfect gap regeneration. A simple gap regeneration model gives good first order estimation of these losses.

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7. References

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